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FATIGUE OF COMPOUND CYLINDERS

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Compound cylinders have historically been designed such that the maximum shear stress is equal in each cylinder sector. This is the optimum condition for yielding in the cylinder since all sectors yield at the same pressure. If compound cylinders are subject to fatigue, this is not the case. A better design criterion is to equate the maximum tensile normal stress in each sector, since the maximum tensile stress controls fatigue crack propagation. This assumes that most of the life is spent in crack propagation. The derivation

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of the geometric conditions resulting in equal tensile stresses in each sector is presented in this report. The geometric optimization of the general cylinder with n sectors is very complex and is a function of the internal and external pressures and the allowable stress, unlike the optimization that assumes that the maximum shear stress dominates. There is one case for which the two methods can be explicitly compared, i.e., a cylinder with only two sectors and no external pressure. This case is studied in detail. The liners of these cylinders are much thinner when the maximum shear stress is assumed than when the maximum shear stress is equalized. To demonstrate that the maximum normal stress is a superior technique, several cylinders were designed to have a constant fatigue life by using both this method and the maximum shear stress method. In all cases, the cylinders had fatigue lives that were at least ten times the design life, the number of cycles at which loading was stopped. However, those cylinders designed according to the maximum normal stress theory showed evidence of crack initiation in many cases. Although both methods give very conservative results, the maximum shear stress theory is considered overly conservative, since it permits much lower internal pressures than the maximum normal stress theory.

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NOMENCLATURE

c	crack length
C_1	material constant
D	the denominator of a derivative
dc/dN	fatigue crack growth rate
ID	inside diameter
k_i	radius ratio of the i^{th} cylinder sector ($k_i=r_{i+1}/r_i$)
K_{Ic}	fracture toughness
K_{max}	maximum stress intensity factor
K_{min}	minimum stress intensity factor
m	material constant
n	total number of sectors in a compound cylinder
N	number of fatigue cycles
OD	outside diameter
p_i	pressure that acts as r_i when the cylinder is pressurized
r_i	i^{th} interface radius
S_{allow}	allowable stress intensity
X	overall radius ratio of a compound cylinder ($X=r_{n+1}/r_1$)
X_i	radius ratio of the $i+1^{th}$ and i^{th} sectors ($X_i=r_{i+2}/r_i = k_{i+1}k_i$)
ΔK_{eff}	effective stress intensity factor range
σ_{max}	maximum normal stress
τ_{max}	maximum shear stress

INTRODUCTION

Several techniques have been used to produce cylinders containing very high pressures. Among these are autofrettage, wire wrapping, and interference fitting of several sectors to produce a multilayered cylinder (ref 1). Historically, these methods have been developed to strengthen cylinders against plastic deformation due to static pressure. If the cylinder is subjected to repeated pressure, it must be designed for optimum fatigue resistance rather than for static strength. In the case of autofrettaged cylinders, the optimum condition for static loading and fatigue loading is essentially the same. This is not the case, however, when designing compound cylinders.

Many investigators have studied compound cylinders and derived the optimum geometric conditions to prevent elastic breakdown (ref 2). It was found that the sectors should be designed such that the maximum shear stress in each sector is equal. Thus, each sector should begin to yield simultaneously. If we consider fatigue failure rather than elastic breakdown, the criterion for the optimum condition would be either for fatigue crack initiation to occur simultaneously in each sector or for pre-existing cracks to grow at the same rate in each segment. For this study, we chose to optimize with respect to crack propagation, assuming that most of the life is expended in crack growth and that cracks initiate very early in the fatigue process.

The effects of mean stress on crack propagation must be taken into account to predict fatigue crack growth in multilayered cylinders, since liners are subjected to compressive residual stress and jackets are subjected to tensile residual stress due to the interference fit of manufacture. Some guidance can be taken from prior work on crack propagation in autofettaged cylinders. Fatigue cracks initiated at the inside diameter (ID) must grow through compressive residual stresses, while cracks initiated at the outside diameter

(OD) grow through a tensile residual stress field. The ID case has been studied more fully and a crack growth model in residual stress fields based on crack closure gives adequate results (ref 3). Crack closure occurs when the applied pressure is insufficient to overcome the compressive residual stresses and open the crack. Fatigue crack growth is assumed to occur only while the crack is indeed open and the effective stress intensity factor range ($\Delta K_{eff} = K_{max} - K_{min}$) which governs crack growth rate (dc/dN) is limited to K_{max}. The power law that represents crack growth under these circumstances is represented as

$$\frac{dc}{dN} = C_1 K_{max}{}^m \tag{1}$$

Although few studies have been made on the tensile residual stress effects in autofrettaged cylinders, some studies have been made where several crack growth laws were used to predict crack growth through the tensile residual stress field in the outer portion of autofrettaged cylinders (ref 4). Tensile residual stresses increase the rate of crack growth rather than retard it as with compressive residual stresses. The application of the model represented in Eq. (1) did not give the best characterization of the crack growth behavior, but it was determined to be the most conservative of those studied. It is not unreasonable to use Eq. (1) for crack growth through tensile residual stress fields as well as compressive residual stress fields.

The optimization of sectored cylinders with respect to fatigue crack growth (accounting for residual stress effects) then involves equating the maximum stress intensity factor that occurs during the pressure cycling in each of the sectors in the multilayered cylinder. This has the added advantage of optimizing with respect to fracture, since final fracture occurs when K_{max} is equal to the fracture toughness (K_{Ic}) of the material from which the cylinder is made.

Therefore, we should be able to design a sectored cylinder in which all the sectors fail at the same number of pressure cycles.

To actually ensure that K_{max} is equal in each cylinder is an involved problem. Stress intensity factor calibrations for thick-walled cylinders with various radius ratios (k) must be known. This is further complicated by the fact that the liner will have pressure in the crack, while the jackets will not. To simplify matters, we assume that the maximum stress intensity factor is proportional to the maximum normal stress ($K_{max} \propto \sigma_{max}$). Then we can proceed with the derivation of optimum geometric designs for various configurations.

ANALYSIS

The Outermost Jackets

Figure 1 shows the schematic of a compound cylinder containing small fatigue cracks of equal length in each sector. The variables used are r_i which denotes the i^{th} interface radius, p_i which is the pressure that acts at r_i when the internal pressure is applied, and k_i which is the radius ratio of the i^{th} sector ($k_i = r_{i+1}/r_i$). We consider the two sectors i and i+1 separately from the remainder of the total cylinder and assume that the local radius ratio of this pair is fixed and given the value X_i, ($X_i = r_{i+2}/r_i$). The cylinder should be constructed so that the maximum normal stress in each sector is equal. The maximum stress is the hoop stress at the inner radius of each sector and is given by the Lamé solution (ref 2).

At r_i:

$$\sigma_{max} = \frac{k_i^2}{k_i^2 - 1} \left(-2p_{i+1} + p_i \frac{k_i^2 + 1}{k_i^2}\right) \tag{2}$$

At r_{i+1}:

$$\sigma_{max} = \frac{k_{i+1}^2}{k_{i+1}^2 - 1} \left(-2p_{i+2} + p_{i+1} \frac{k_{i+1}^2 + 1}{k_{i+1}^2}\right) \tag{3}$$

To determine the optimum geometric conditions for equal stresses, we solve both Eqs. (2) and (3) for the pressure acting at the interface between the i^{th} and the $i+1^{th}$ sectors (p_{i+1}). Since p_{i+1} is the same in both equations, we equate expressions for p_{i+1} and solve the resulting equation for σ_{max}

$$\sigma_{max} = p_i \frac{k_i^4 + (X_i^2+1)^2 k_i^2 + X_i^2}{-k_i^4 + (3X_i^2-1)k_i^2 - X_i^2} - 2p_{i+2} \frac{4X_i^2 k_i^2}{-k_i^4 + (3X_i^2-1)k_i^2 - X_i^2} \qquad (4)$$

The optimum mating of k_i and k_{i+1} for a fixed X_i occurs when we set either $d\sigma_{max}/dk_i$ or $d\sigma_{max}/dk_{i+1}$ equal to zero. Equation (4) was derived such that the differentiation is with respect to k_i. It could also have been developed for differentiation with respect to k_{i+1}, however, the end result would be the same since k_i and k_{i+1} are related to X_i by $X_i = k_i k_{i+1}$. Since Eq. (4) is a fraction, its derivatives are also fractions. The condition for the derivative being equal to zero is that the numerator of the derivative is equal to zero, namely

$$D \frac{d\sigma_{max}}{dk_i} = 8X_i^2 k_i (k_i^4 p_i + X_i^2 (p_{i+2} - p_i)) = 0 \qquad (5)$$

where D is a function of k_i and the denominator of the derivative of Eq. (4). It is clear that the ideal mating of these two adjacent sectors is not solely a function of geometry, but is also a function of the boundary conditions p_i and p_{i+2}. The roots of Eq. (5) can be determined by the change of variable $k_i' = k_i^2$. This reduces the quartic to a quadratic whose roots are determined by applying the quadratic formula. The optimum value of k_i is taken as the square root of the positive root of k_i'

$$k_i = \sqrt{X_i} \left(1 - \frac{p_{i+2}}{p_i}\right)^{1/4} \qquad (6)$$

If the pressure p_{i+2} is equal to zero, Eq. (6) reduces to the same solution that results when the geometry is optimized by equating the maximum shear stress in each sector (ref 2), namely, $k_i = k_{i+1} = \sqrt{X_i}$. This is the case for the two outermost sectors when the total cylinder is subjected to only internal pressure and there are more than two sectors. If there are only two sectors, the following optimization must be used that includes the effects of internal pressure in the crack.

The Liner and Inner Jackets

The above derivation assumes that no pressure acts in any crack. The innermost liner is subjected to the applied fluid pressure. If the crack is open, pressure will infiltrate the crack and act on the crack faces. The resulting maximum stress intensity factor will be proportional to the maximum normal stress from the Lamé solution plus the internal pressure. A special derivation must be followed to account for this. The stresses that are to be equated now are

At r_1:

$$\sigma_{max} = \frac{k_1^2}{k_1^2 - 1} (-2p_2 + 2p_1) \tag{7}$$

At r_2:

$$\sigma_{max} = \frac{k_2^2}{k_2^2 - 1} \left(-2p_3 + p_2 \frac{k_2^2 + 1}{k_2^2}\right) \tag{8}$$

The optimum value of k_1 is determined by eliminating the intermediate pressure p_2 and solving for σ_{max}, as above. In this instance, the resulting equation as a function of k_1 and X_1 (which is again assumed to be fixed) is

$$\sigma_{max} = p_1 \frac{2X_1^2k_1^2 + 2k_1^4}{-k_1^4 + (3X_1^2-1)k_1^2 - X_1^2} - p_3 \frac{4X_1^2k_1^2}{-k_1^4 + (3X_1^2-1)k_1^2 - X_1^2} \tag{9}$$

Once more the maximum stress value is not a function of geometry alone, but is also a function of the boundary conditions p_1 and p_3. The optimum geometry is found by differentiating with respect to k_1 and setting the derivative equal to zero. Again, the result is a quartic

$$0 \frac{d\sigma_{max}}{dk_1} = 4k_1([(4X_1^2-1)k_1^4 - 2X_1^2k_1^2 - X_1^4]p_1 - (2X_1^2k_1^4 - 2X_1^4)p_3) \qquad (10)$$

The optimum value of k_1 is determined by reducing the quartic to a quadratic and using the quadratic formula in the same manner as outlined above.

$$k_1^2 = \frac{X_1^2 \pm X_1^2\sqrt{1 + [4X_1^2 - 1 + 2(p_3/p_1)(1-10X_1^2) - (p_3/p_1)^2]}}{4X_1^2 - 1 - 2X_1^2(p_3/p_1)} \qquad (11)$$

Equation (11) is a complicated expression showing that the optimum geometry is a function of the pressures p_1 and p_3.

To optimize the mating of the general multilayered cylinder with n sectors and an overall radius ratio of $X = k_1 \cdot k_2 \cdots k_i \cdots k_{n-1} \cdot k_n$ subjected to internal pressure p_1 and external pressure p_{n+1}, is indeed a difficult process. The solution is obtained by starting at the OD or the ID where the value of p_{n+1} or p_1 is known and working either in toward the ID or out toward the OD, whichever the case may be. For example, if we start at the OD and p_{n+1} is known, we can use Eq. (4) to solve for the ratio p_{n+1}/p_{n-1} in terms of σ_{max}, the local combined radius ratio X_{n-1}, and k_{n-1} which can be substituted into Eq. (6) to give an expression for the optimum k_{n-1} in terms of X_{n-1}. This can then be substituted into either Eq. (2) or (3) to determine an implicit expression for p_{n-1}. Repeating the procedure for the mating pair of k_{n-2} and k_{n-1} and so on until we get to the pair k_2 and k_1, where we must use Eqs. (7), (8), (9), and (11) rather than Eqs. (2), (3), (4), and (6), we generate a system of n-1 nonlinear equations involving 2n-1 variables: $k_1, \cdots k_n$ and $X_1, \cdots X_{n-1}$. We add another n-1

equation by realizing that $X_i = k_i k_{i+1}$ and the final equation is the design parameter for X

$$X = k_1 \cdot k_2 \cdots k_i \cdots k_{n-1} \cdot k_n \tag{12}$$

Thus, in theory it is possible to optimize the general multilayered cylinder given the requirements: X, p_1, n, p_{n+1}, and an allowable σ_{max}. This is not a trivial task as some technique is required to solve systems of nonlinear simultaneous equations. The actual solution method depends on the particular case and is left to the reader. The important thing to note about the results obtained is that unlike the case for optimizing with respect to maximum shear stress, there is no single geometric ideal mating based on X alone. The optimum mating of the general compound cylinder for fatigue application is a function of the allowable σ_{max} and the design pressure. In other words, for a given X, the optimum design for an allowable stress of 500 MPa and 20 kbar internal pressure would be different than the design for an allowable stress of 600 MPa and 15 kbar internal pressure. If the cylinder were designed such that the optimum mating for either of the two design conditions stated above was based on the shear stress criterion, the geometry of each sector $k_1 \cdots k_n$ would be the same. Only the interface pressures would be different to meet the requirements.

The Special Case of Two Sectors and No External Pressure

There is one case that can be optimized explicitly and is a function of geometry alone. This occurs if $n = 2$ and $p_3 = 0$. We use this case to compare the optimization method developed here and the more traditional approach of equating maximum shear stresses. The ideal mating for a required radius ratio X and two sectors is given by

$$k_1 = \sqrt{X} \tag{13}$$

$$k_1^2 = \frac{X^2 + 2X^3}{4X^2 - 1} \tag{14}$$

Equation (13) is from the shear stress criterion (ref 2) and Eq. (14) is Eq. (11) simplified by setting p_3 equal to zero. It is clear that geometric optima based on the two different criteria are substantially different. The two equations are plotted in Figure 2 for a clearer comparison. The liner is always thinner when we use the maximum normal stress criterion because of the necessity for a larger compressive residual stress at the bore than in the case of equal shear stresses.

A more interesting comparison is between the allowable stresses. Suppose we design a cylinder for a given fatigue life and find that the allowable stress intensity is S_{allow}. Using the normal stress criterion, the design pressure is found from $\sigma_{max} = S_{allow}$, and using the shear stress criterion, the design pressure is found from $2\tau_{max} = S_{allow}$. The equations for the allowable pressure are

Shear stress criterion:

$$\frac{p}{S_{allow}} = \frac{X - 1}{X} \tag{15}$$

Normal stress criterion:

$$\frac{p}{S_{allow}} = \frac{-k_1^4 + (3X^2-1)k_1^2 - X^2}{2X^2k_1^2 + 2k_1^4} \tag{16}$$

Equations (15) and (16) are plotted in Figure 3. This graph shows that for a given X and S_{allow}, the normal stress criterion will always allow for a greater design pressure than if the shear stress criterion were applied. This demonstrates that if the initial assumptions are correct, and the maximum normal stress dominates the fatigue behavior of compound cylinders, the use of the maximum shear stress criterion is not the optimum procedure. Furthermore, if the cylinder were designed according to the shear stress criterion, those cylinders would be quite overdesigned.

EXPERIMENTAL PROCEDURE

To demonstrate that designing sectored cylinders for fatigue application using the shear stress optimization is overly conservative, a short experimental program was devised. It was decided to design cylinders for a given fixed X of 2.25 and a given fatigue life of either 20,000 cycles or 100,000 cycles. The material used was ASTM A-723 Grade 3, Class 1 pressure vessel steel. This steel has a minimum tensile strength of 795 MPa. Using the fatigue (S - N) curves for this material that appear in the ASME Pressure Vessel Code, Section VIII, Division 2, we find that the allowable stress intensity amplitude is 261 MPa for a life of 20,000 cycles and 174 MPa for a life of 100,000 cycles.

Using Eqs. (15) and (16) we can determine the applied internal pressure that results in the allowable stress intensity amplitudes. These are presented in Table I along with the other pertinent data necessary to produce two sets of specimens (one each for the two different design methods). The additional data include liner radius ratios k_1 and interference pressures for each design. For each condition, two specimens were manufactured and fatigue tested using Bristol high pressure fatigue machines.

TABLE I. DESIGN PARAMETERS FOR THE EXPERIMENTAL STUDY - OVERALL RADIUS RATIO X = 2.25

Design Life (Cycles)	Optimization Criterion	k_1	Interference Pressure (MPa)	Internal Pressure (MPa)
20,000	Shear Stress	1.500	60.7	284.3
100,000	Shear Stress	1.500	35.6	190.3
20,000	Normal Stress	1.203	59.4	370.0
100,000	Normal Stress	1.203	39.8	247.7

RESULTS AND DISCUSSION

The measured fatigue lives of the eight specimens tested are presented in Table II. Pressure cycling of any specimen was suspended once it achieved a life equal to ten times the design life. As evident in Table II, every specimen

TABLE II. MEASURED FATIGUE LIVES COMPARED WITH PREDICTED LIVES

Specimen No.	Optimization Criterion	Design Life (Cycles)	Actual Life (Cycles)
S20-1	Shear Stress	20,000	>200,000
S20-2	Shear Stress	20,000	>200,000
S100-1	Shear Stress	100,000	>1,000,000
S100-2	Shear Stress	100,000	>1,000,000
N20-1	Normal Stress	20,000	>200,000
N20-2	Normal Stress	20,000	>200,000
N100-1	Normal Stress	100,000	>1,000,000
N100-2	Normal Stress	100,000	>1,000,000

tested met this criterion. Some specimens designed according to the maximum normal stress criterion had cracked liners, whereas no specimen designed according to the maximum shear stress criterion showed any evidence of crack initiation when cycling was stopped. Although some liners were cracked, there were no gross, through-thickness failures. Therefore, all specimens had a safety factor of ten on design fatigue life. At first glance, this may seem to prove little, except that the maximum shear stress criterion and the maximum normal stress criterion are both very conservative. The significant result from these tests is that the maximum shear stress criterion is clearly more conservative than the maximum normal stress criterion. For a fixed overall radius ratio X and allowable stress intensity S_{allow}, the maximum normal stress criterion permits substantially more pressure to be safely contained. In the

particular case of X = 2.25 studied here, the permitted pressure is 30 percent greater using the maximum normal stress than the maximum shear stress. On the other hand, if we design a cylinder for a given internal pressure and allowable stress intensity, then the two theories will allow very different cylinders as a result. Let us assume that the ratio of internal pressure to allowable stress intensity is 0.6. From Figure 3, we need a cylinder with X = 2.5 using the maximum shear stress theory, while using the maximum normal stress theory a cylinder with an overall radius ratio of X = 1.9 is necessary. Since this is the case in many instances, rather than designing to a fatigue life requirement, using the maximum normal stress theory developed herein is more advantageous since much less material is required.

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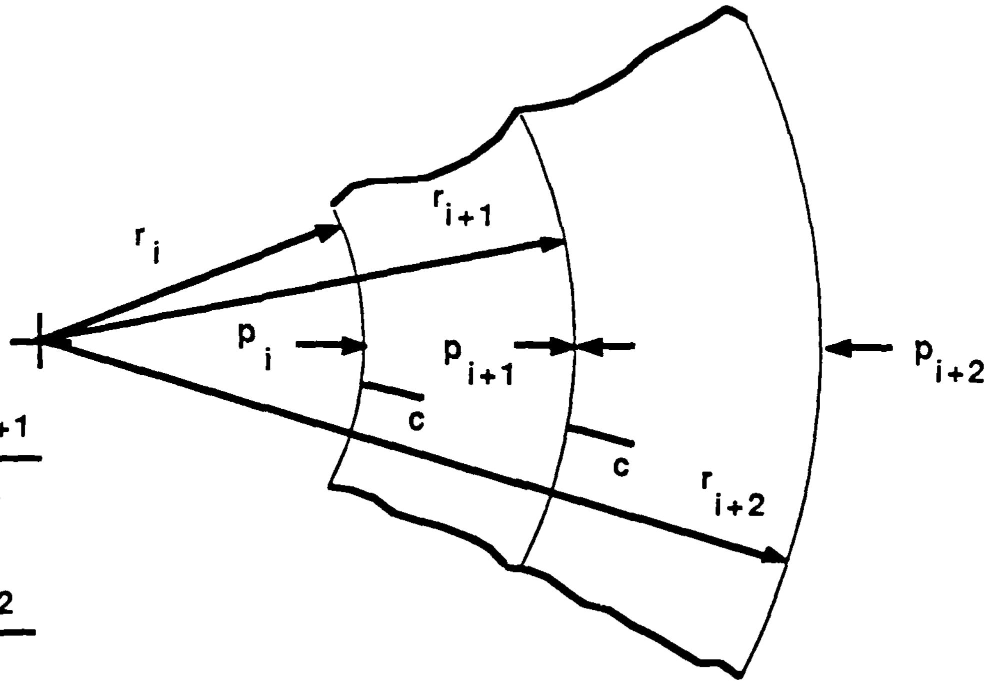

$$k_i = \frac{r_{i+1}}{r_i}$$

$$X_i = \frac{r_{i+2}}{r_i}$$

Figure 1. Schematic of a sectored cylinder.

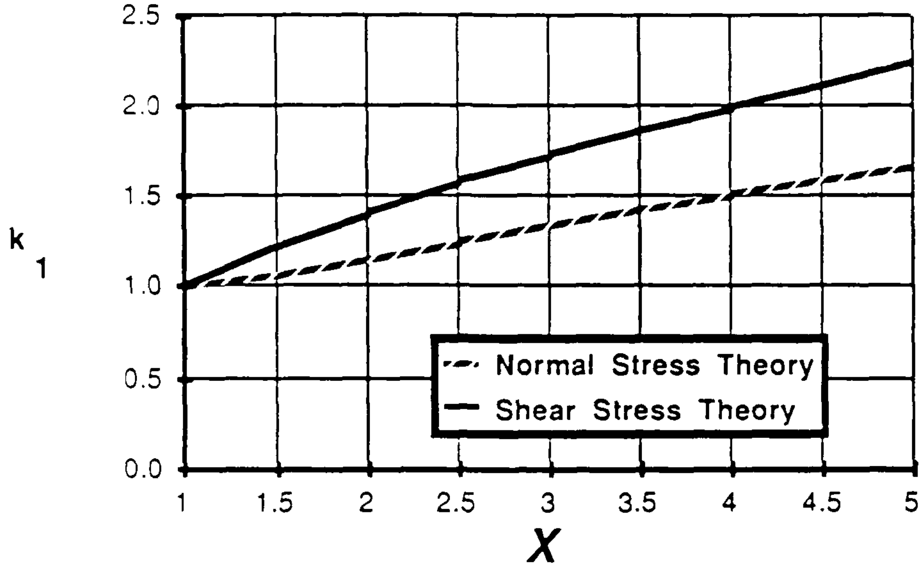

Figure 2. Comparison of ideal liner matings for two theories.

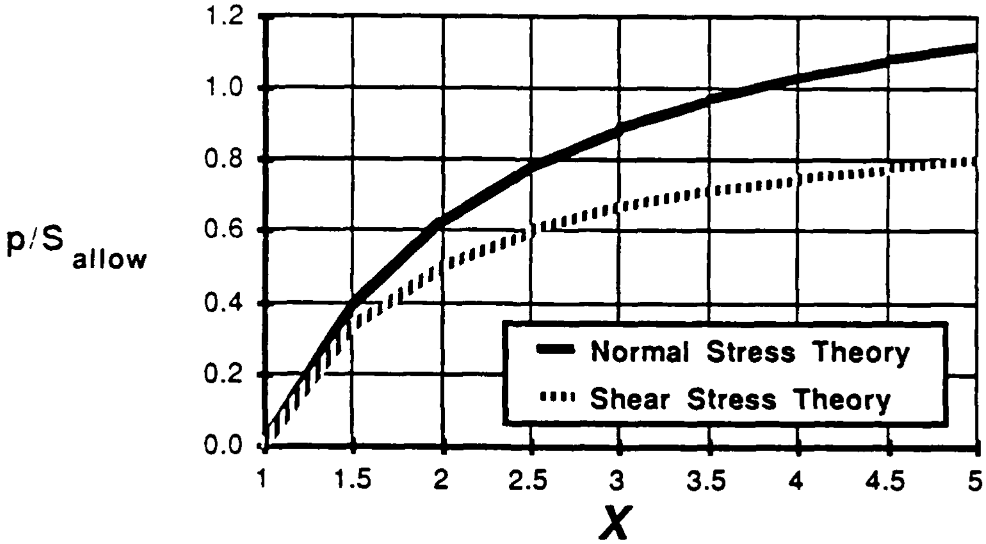

Figure 3. Comparison of allowable pressures for two theories.

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